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CALCULATING HEAT TRANSFER IN DIATHERMAL SYSTEMS IN THE SUPERCRITICAL BOILING REGIME

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CALCULATING HEAT TRANSFER IN DIATHERMAL SYSTEMS IN THE SUPERCRITICAL BOILING REGIME

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ABSTRACT. Certain freatures of the heat transfer of diathermal systems and their analysis for the critical and supercritical boiling regimes are examined. A diathermal heat transfer scheme, when $D \neq 0$, is studied.

Severe requirements with regard to specific thermal loads, capacity, effec- /89* tiveness, and reliability are imposed on present-day heat exchange equipment, and engine cooling systems, whose heat-removing fluids are high temperature heat-carrying agents.

Generally accepted heat transfer schemes and methods for calculating the heat exchange have now been developed. However, they are all based on the principle of heat transfer through the structural wall by thermal conduction alone, when the convective q_c and radiative q_r heat fluxes are absorbed completely by the surface of the wall, which is considered to be athermal, i.e., $D = 0$ and $A + R = 1$. In this connection very large thermal fluxes, reaching the critical values (q_{cr}), are created on the heat-removing surface of the equipment.

Further intensification of the heat exchange by boiling of the heat-absorbing (cooling) fluid is difficult.

In contrast with the existing schemes, we shall consider the diathermal heat transfer scheme, when $D \neq 0$. It is assumed that the heat exchangers or engine combustion chamber and nozzle in question operate at very high working

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* Numbers in the margins indicate pagination in the original foreign text.

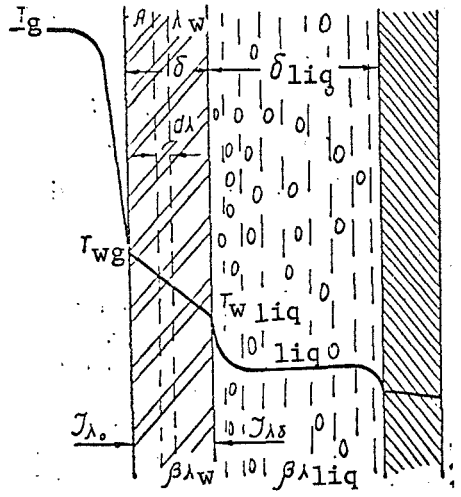

Figure 1. Diagram for analysis of heat transfer in a diathermal wall.

fluid temperatures of the order of $2 \cdot 10^3 - 6 \cdot 10^3$°K and above, when the specific radiant heat fluxes play the dominant role in the heat exchange [5, 6].

The overall specific thermal fluxes q_Σ corresponding to these temperatures are very high and amount to millions of kilocalories per square meter per hour. In such cases the heat exchange proceeds with boiling of the heat-transfer agent, and in the case of further increase of thermal density, in the case of forced augmentation of the heat exchange, the nucleate boiling regime may transition into the film regime with the associated consequences [2, 3].

The critical regime is also of interest from the viewpoint of establishing the maximal possible thermal heads. We shall examine certain peculiarities of the heat transfer of diathermal systems and their analysis for the critical and supercritical boiling regimes.

Just as in the athermal systems, the working fluid with the temperature T_g acts on the wall surface (Figure 1), giving up the thermal energies q_c and q_r

$$q_c = \alpha_g (T_g - T_{w.g}),$$

$$q_r = \varepsilon_w \cdot c_0 \left[\varepsilon_g \left(\frac{T_g}{100} \right)^4 - A_g \left(\frac{T_{w.g}}{100} \right)^4 \right].$$

However, not all the heat $q_\Sigma = q_c + q_r$ is absorbed by the wall surface; /90
only q_c is absorbed, and by virtue of the diathermal nature of the wall only the portion $q_r(1 - D)$ is absorbed by the wall through its entire thickness [1, 7].

It is known that when thermal rays with the spectral intensity I_λ pass through a diathermal body their intensity falls off in proportion to the wall radiant energy absorption coefficient $\beta_{\lambda w}$, corresponding to the given wavelength λ, i.e.,

$$dI_\lambda = -\beta_{\lambda w} \cdot I_\lambda \cdot d\delta,$$

where $d\delta$ is an elementary layer of the diathermal wall; dI_λ is the radiation intensity decrease.

Integrating over the entire wall thickness δ

$$\int_0^\delta \frac{dI_\lambda}{I} = \int_0^\delta -\beta_{\lambda w} \cdot d\delta,$$

we have

$$\ln \frac{I_{\lambda\delta}}{I_{\lambda_0}} = -\beta_{\lambda w} \cdot \delta, \quad \frac{I_{\lambda\delta}}{I_{\lambda_0}} = e^{-\beta_{\lambda w} \cdot \delta}$$

or

$$\frac{I_{\lambda_0} - I_{\lambda\delta}}{I_{\lambda_0}} = 1 - e^{-\beta_{\lambda w} \cdot \delta} = A\lambda.$$

But, since A + R + D = 1, and setting R = 0, for

$$D = \frac{E'}{E} = \frac{\int_{\lambda=a}^{\lambda=b} I\lambda \cdot d\lambda}{\int_{\lambda=0}^{\lambda=\infty} I_\lambda \cdot d\lambda} \quad [5]$$

we obtain the equation relating the energies absorbed by and passing through the diathermal wall as a function of δ, β, and $\Delta\lambda$:

$$\frac{\int_{\lambda=a}^{\lambda=b} I_\lambda \cdot d\lambda}{\int_{\lambda=0}^{\lambda=\infty} I_\lambda \cdot d\lambda} + \int_{\lambda=0}^{\lambda=\infty} (1 - e^{\beta_{\lambda w} \cdot \delta})\, d\lambda - 1 = 0. \qquad (1)$$

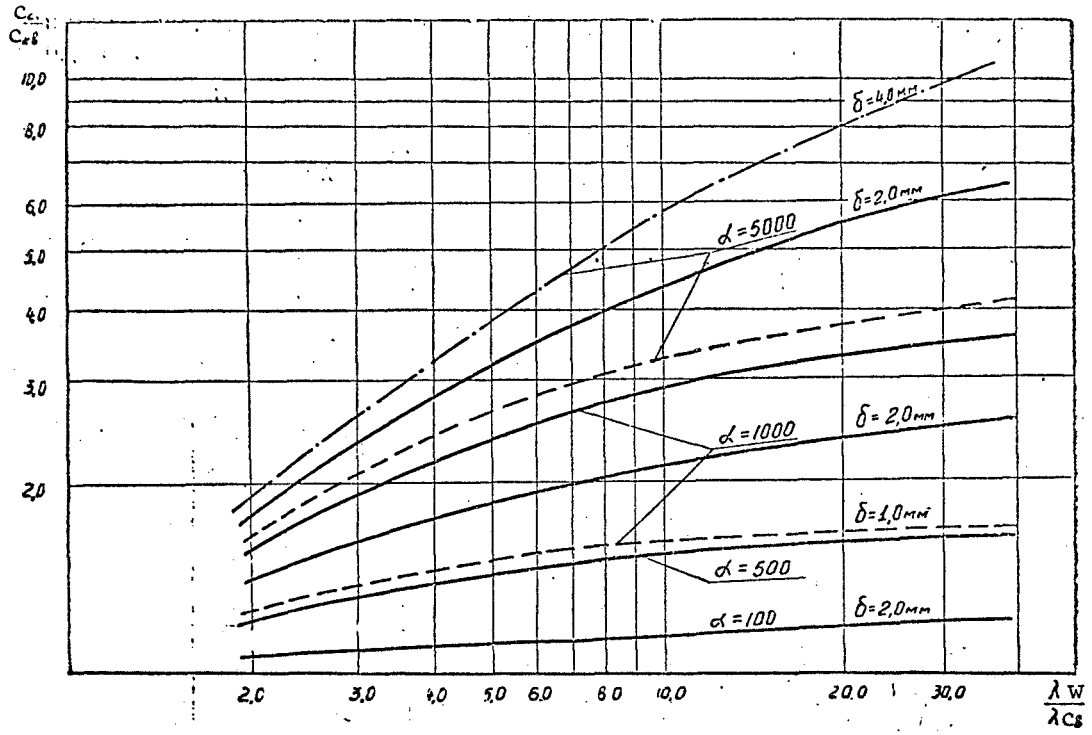

Figure 2. Effect of wall heat conductivity on the heat transfer coefficient; δ is the wall thickness.

Thus, the following amount of heat is transmitted through the wall to its heat-removal surface by thermal conduction

$$q_g \int_{\lambda=0}^{\lambda=\infty} (1 - e^{-\beta_\lambda w \cdot \delta}) d\lambda + q_c = q_\lambda. \qquad (2)$$

At the same time, this must be less than q_{cr}; otherwise, as the heat is transferred to the liquid coolant, the nucleate boiling will transition into film boiling with a decrease of α_{liq}. /91

The wall thermal resistance δ/λ_w is of considerable importance in heat passage through the wall. The reason is that the majority of the diathermal materials have quite low values of the thermal conductivity λ_w in comparison with the metals, which at first glance might seem to interfere markedly with the heat transfer through the wall. However, we see from Figure 2 that with an increase of the wall thermal resistance the heat transfer as a whole decreases by nearly an order of magnitude less, since the primary obstacle to heat transfer is the thermal resistance of the thermal boundary layer, approximated by

$1/\alpha_g$. Only for very large α_g of the order of $(2 - 5) \cdot 10^3$ W/m^2 C° do the thermal resistances δ/λ_w and $1/\alpha_g$ become commensurate.

In other words, the heat transfer is restricted not by the decrease of λ_w but by the significantly smaller values of α_g; therefore the apprehension about the reduction of the overall heat transfer coefficient K is not justified.

Examining and analyzing the critical regimes alone, it is easy to see that increase of the wall thermal resistance cannot in any way affect the critical boiling mechanism itself, and therefore has no effect on q'_{cr}. /92

The further heat transfer takes place at the wall-liquid boundary.

In the thermal boundary layer of the liquid the heat is transferred from the wall to the liquid by convection

$$q_{r\,\overline{liq}} = \alpha_{liq}(T_{w\,liq} - T_{liq})$$

and also by the thermal radiation absorbed by the liquid, which has the absorptivity $\beta_{\lambda\,liq}$. The radiation is absorbed up to the layer depth δ_{liq}, determined from the equation

$$q_{r\,\overline{liq}} = q_r \int_{\lambda=0}^{\lambda=\infty} (1 - e^{-\beta_\lambda \delta_{liq}}) d\lambda.$$

If $q_{c\,liq}$ approaches q_{cr} in magnitude, the liquid absorbs a supercritical /93 amount of heat, equal to

$$q'_{cr} = \alpha_{liq}(T_{w\,liq} - T_{liq}) + q_r \int_{\lambda=0}^{\lambda=\infty} (1 - e^{-\beta_{\lambda liq}\delta_{liq}}) d\lambda. \quad (3)$$

In order to evaluate the supercritical boiling regimes it is advisable to introduce a dimensionless quantity which characterizes the degree of heat transfer intensification and represents the ratio of the thermal fluxes characteristic of the critical thermal regime in diathermal systems (called here the supercritical regimes) to the critical fluxes in the conventional athermal systems

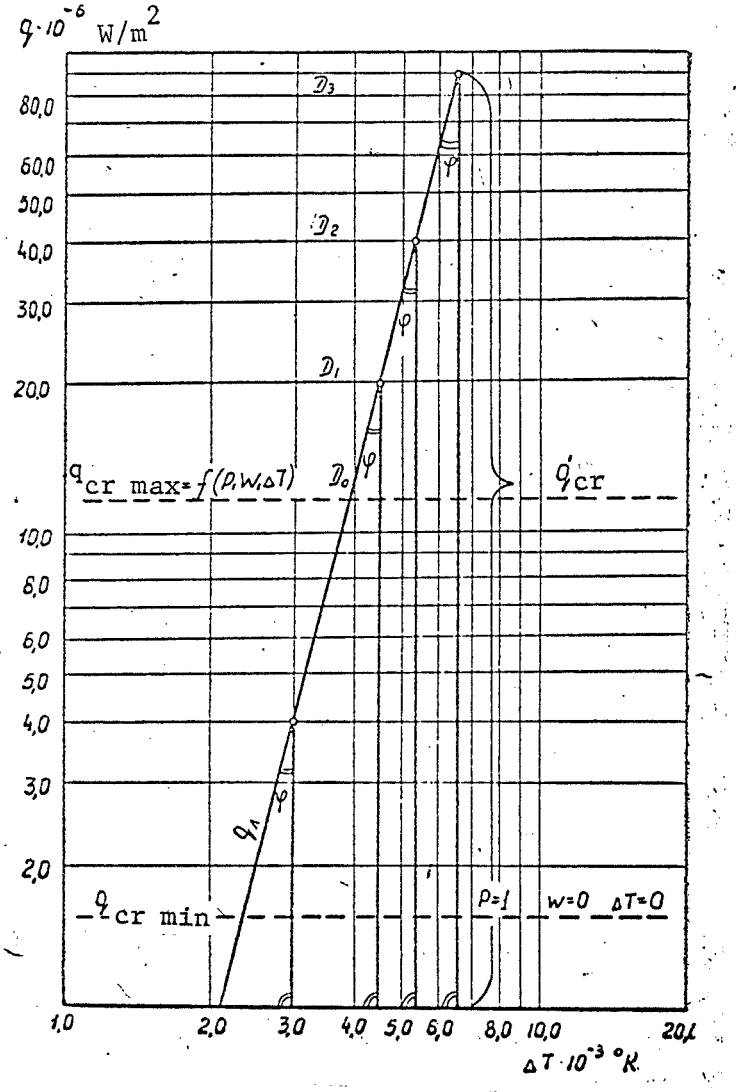

Figure 3. Diagram for analysis of the heat transfer intensification coefficient K_i.

Hence

$$q'_{cr} = \frac{\Delta T}{\operatorname{tg}\varphi},$$

but

$$q'_{cr} = q'_{cr\ r} + q_c,$$

$$K_{\text{II}} = \frac{q'_{cr}}{q_{cr}}. \qquad (4)$$

We note that with increase of α_g the possibilities for heat transfer intensification diminish and K_i decreases.

The effect of α_g on the supercritical regime and the quantitative relationships between α_g and K_i may also be seen from Figure 3, where q_r, q_c, and q_{cr} are plotted in the logarithmic coordinates $q \cdot \Delta T$.

Since q_r is represented by a straight line forming the angle ϕ with q_{cr} for ϕ = const, connections between the parameters may be established for all the heat transfer regimes.

Figure 3 implies that

$$\operatorname{tg}\varphi = \frac{\Delta T}{q'_{cr}}.$$

while

$$q_{cr} = q_c + q_r(1 - D)$$

and

$$K_и = \frac{q'_{cr}}{q_{cr}} = \frac{q_{cr\,r} + q_c}{q_{cr}} = \frac{\frac{\Delta T}{\operatorname{tg}\varphi} + q_c}{q_c + q_r(1 - D)},$$

or

$$q_c = \frac{\frac{\Delta T}{\operatorname{tg}\varphi} - K_i \cdot q_r(1 - D)}{K_i - 1},$$

hence

$$\alpha_g = \frac{\frac{\Delta T}{\operatorname{tg}\varphi} - K_i \cdot q_r(1 - D)}{(K_i - 1)\,\Delta T}. \qquad (5)$$

Since T_g is given, q_r, D, and ΔT are known and uniquely related with T_g, while the angle ϕ is constant for all the regimes [4]. Then for ε = const, α_g = = $f(k_i)$.

Transforming (5), we obtain the equation relating α_g with K_i in the form /94

$$\alpha_g = \frac{q_r[1 - K_i(1 - D)]}{(K_i - 1)\,\Delta T}. \qquad (6)$$

If we now plot these analytic relations, we see quite clearly and can evaluate quantitatively the spheres of possible thermal regimes: subcritical, critical, supercritical, and hypercritical.

In Figures 4, 5, and 6 the subcritical regimes lie in the regions below ab and a, respectively, in which the conditions must be satisfied

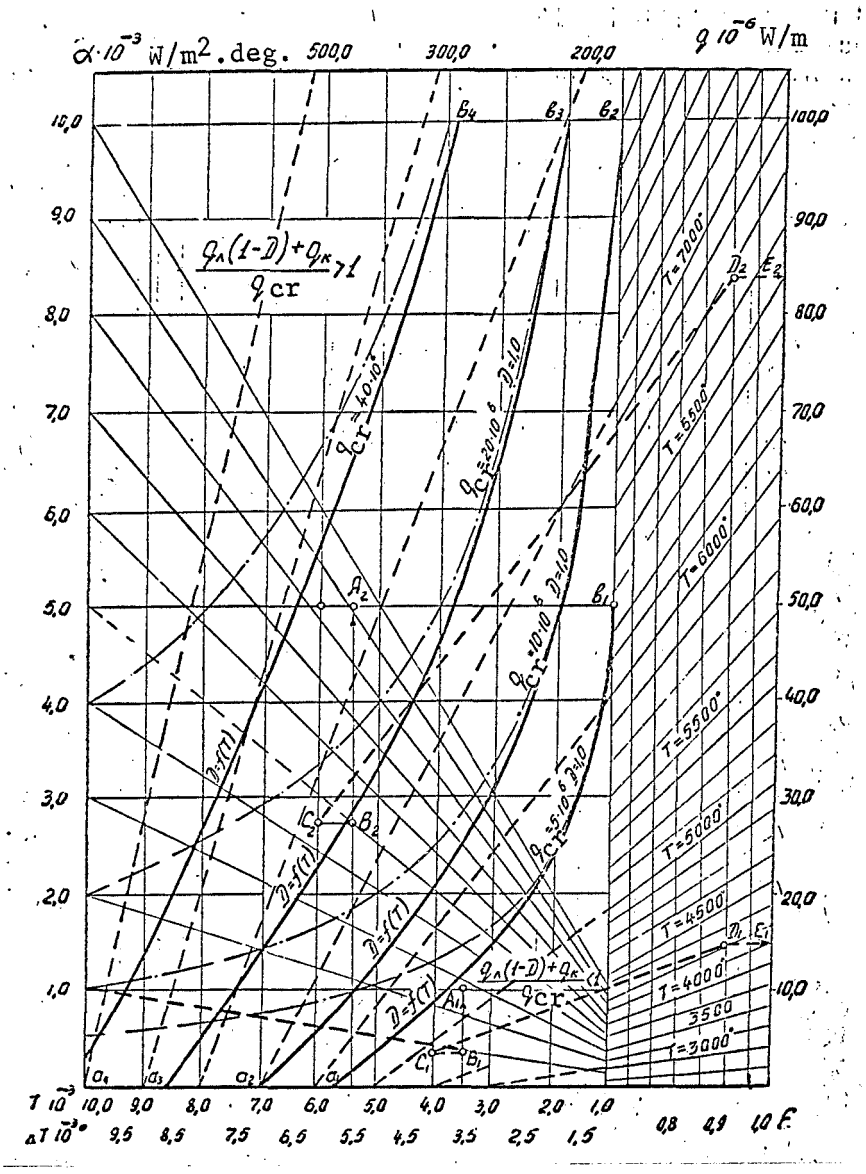

Figure 4. Nomogram for determining q'_{cr}.

$$\frac{q_r(1-D)+q_c}{q_{cr}} = \frac{\varepsilon_w c_0 \left[\varepsilon_g \left(\frac{T_g}{100} \right)^4 - A_g \left(\frac{T_{w.g}}{100} \right)^4 \right] (1-D) + \alpha_g (T_g - T_{w.g})}{K_w \cdot r \sqrt{g\gamma''} \sqrt[4]{\sigma(\gamma' - \gamma'')} \left[1 + 0{,}065 \left(\frac{\gamma'}{\gamma''} \right)^{0,8} \cdot \frac{c \cdot \Delta T}{g} \right]} < 1,$$

where r is the latent heat of vaporization,

γ' is the liquid density,

γ'' is the vapor density,

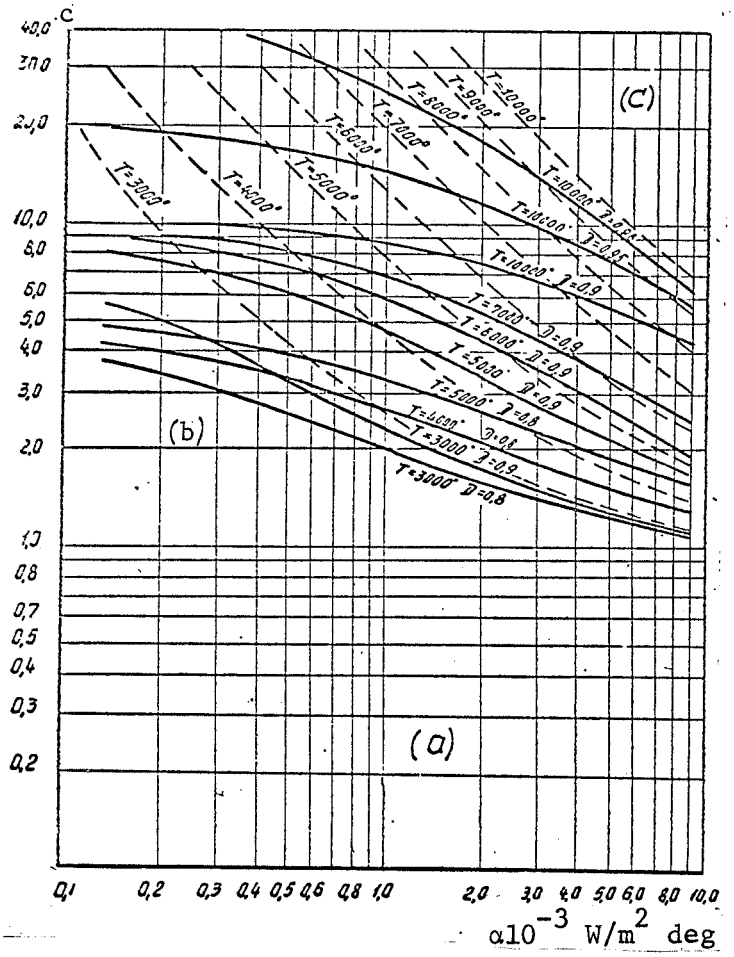

Figure 5. K_i versus α_g and T_g.

c is the liquid heat capacity, σ is the surface tension coefficient.

The curves a_1b_1, a_2b_2, and so on correspond to the critical regimes.

The supercritical regimes lie in the regions above ab and b, respectively, where

$$\frac{q_g(1-D)+q_c'}{q_{cr}} > 1,$$

and the hypercritical regime lies in zone c (Figures 5 and 6), where the boiling regime is obviously of the film type, in spite of the diathermal heat-transfer scheme.

The critical heat fluxes characteristic of liquid heat-transfer agents for optimal values of P, W, and ΔT are shown by the dot-dash curves in Figure 4. The continuous curves show the same q_{cr}, but with account for the diathermal nature of the wall, equal to $q_c + q_r(1 - D)$. Above these curves lie regions of the forbidden thermal regime, film boiling; below lies the working region for the given heat-transfer agent.

The plot has been shifted twice with respect to α_g. In the first case the region of acceptable or unacceptable heat transfer regimes is determined. Its location is found at the intersection of the horizontal line α_g with the vertical line ΔT. If the resulting point A is above $q_{cr\ cool}$ then the boiling crisis has already occurred as a result of convection alone. In this case there is no point in accounting for the radiant thermal flux, since the boiling regime is

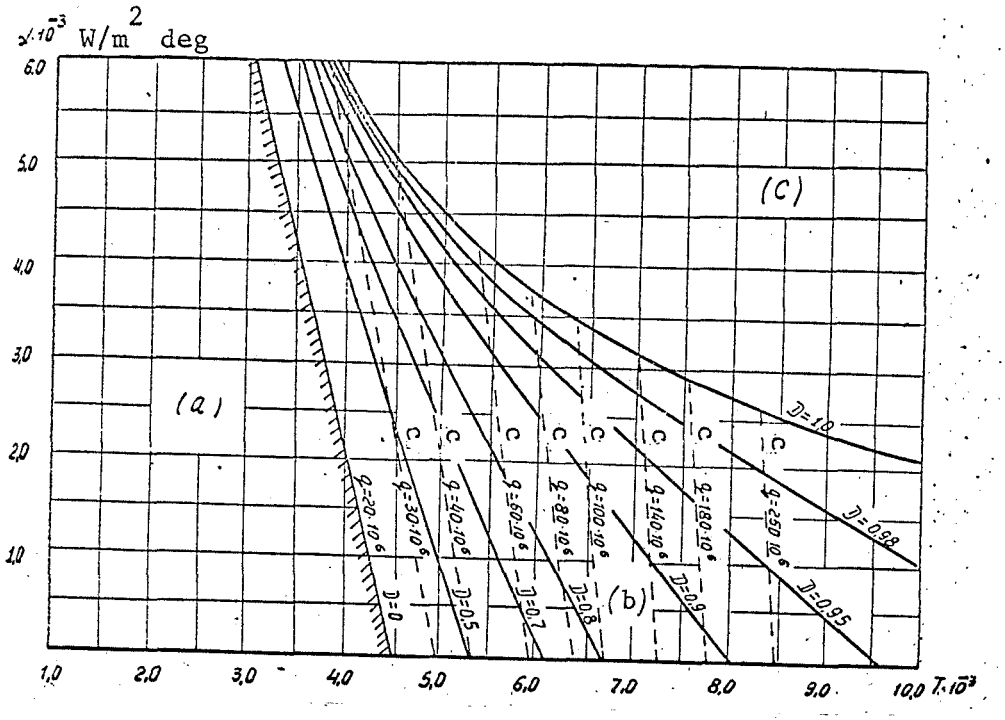

Figure 6. Nomogram for determining acceptable thermal regimes for $q_{cr\ cool} = 20 \cdot 10^6$ W/m^2.

obviously of the film type, i.e., unacceptable. In the second case, when the point A is below $q_{cr\ cool}$, heat transfer is possible; the supercritical boiling regime can be realized.

In order to define this regime, we find the point B at the intersection of the sloping ray α_g and the vertical line ΔT. But this point characterizes only the convective heat transfer, i.e., $q_c = \alpha_g \cdot \Delta T$. In order to obtain the total heat flux we must follow along the horizontal to the intersection at the point C with the isotherm T_g. Then from the dashed ray, whose slope corresponds to /95
the given temperature, with correction for ε, we find the sought thermal density q_Σ and from the graph in Figure 5 we find K_i with account for the diathermal nature of the wall.

We see from Figure 4 that with an increase of $q_{cr\ cool}$ which is acceptable to the coolant and reduction of α_g more extensive possibilities show up for intensification of the heat transfer and increase of K_i.

We must also bear in mind that the nomogram in Figure 4 makes it possible to determine the magnitude of the maximal possible intensification of the heat transfer, but to do this we need the condition for which the characteristic critical thermal densities $q_{cr\ cool}$ of the liquid coolants would coincide in magnitude with the convective component for D = 1.0.

As an example let us consider a heat exchanger made from diathermal elements like quartz, or some other material with analogous spectral diathermance. /96
The heat exchanger surface is exposed to a gaseous flow having the emissivity $\varepsilon = 0.9$, temperature $T_g = 4000°K$ and $\Delta T = 3500\ °K$. The heat transfer coefficient from the gas to the wall $\alpha_g = 1000\ W/m^2$ deg, D=0.8, the liquid coolant has $q_{cr\ cool} = 5 \cdot 10^6\ W/m^2$.

In Figure 4 we find the point A_1 corresponding to zone $\frac{q_r(1 - D) + q_c}{q_{cr\ cool}} < 1$,

i.e., below the line a_1b_1. This means that the heat transfer regime is possible.

Let us find q'_{cr}. We use α_g and ΔT to find the point B_1. Following the horizontal line to intersection with $T_g = 4000°K$, we obtain the point C_1. Following the line parallel to the temperature ray $T_g = 4000°K$, we find the point D_1, and with account for ε we find the point E_1, to which there corresponds /97
$q'_{cr} = 14.5 \quad 10^6\ W/m^2$. On the other hand, from the nomogram of Figure 5 and

the given T_g, α_g, and D = 0.8, we find $K_i = 2.7$ and $q_{cr\ cool} = \frac{q'_{cr}}{K_i} = 5.4 \cdot 10^6$ W/m^2.

Thus, the liquid coolant having the characteristic $q_{cr\ cool} = 5.4 \cdot 10^6\ W/m^2$ can be used in heat exchangers with a thermal density of 14.5 W/m^2 without danger of failure. However, if this capability is not realized, i.e., there is no need to increase the specific thermal fluxes, then this effect makes it possible to improve the reliability of the heat exchangers, which shows up in more stable nucleate boiling without the risk of the occurrence of the film boiling regime.

Let us consider another case in which $T_g = 6000°K$, $\Delta T = 5500°K$, $\alpha_g = 5000$ W/m^2 deg, D = 0.9 and $q_{cr\ cool} = 20 \cdot 10^6\ W/m^2$. We find the point A_2

corresponding to the zone $\frac{q_r(1 - D) + q_c}{q_{cr\ cool}} > 1$; it lies above a_3b_3; therefore the operating regime is not acceptable.

However, if for the same T_g, $T_{w\ g}$, α_g and ε we replace the liquid coolant with another having $q_{cr\ cool} = 40 \cdot 10^6$ W/m^2, the point A_2 lies below a_4b_4, i.e., outside the unacceptable zone, and the heat transfer regime becomes possible.

Let us determine q'_{cr}. We find the point B_2; then we find C_2 and follow /98 the ray corresponding to $T_g = 6000°K$ to the point D_2, and with account for ε we find the point E_2, where $q'_{cr} = 88.5 \cdot 10^6$ W/m^2.

Returning to Figure 5, we obtain $K_i = 2.65$ and $q_{cr\ cool}$. Thus, for $q_{cr\ cool} = 33.4 \cdot 10^6$ W/m^2 the thermal density is acceptable up to $88.5 \cdot 10^6$ W/m^2.

Using (3), (4), and (6), we represent the relationship between T_g, $T_{w\ g}$, α_g, $q_{cr\ cool}$, K_i, and D in the form of a nomogram (Figure 6). Here we show one particular case of heat transfer with a liquid coolant with $q_{cr\ cool} = 20 \cdot 10^6$ W/m^2. The nomogram makes it possible to calculate graphically the heat transfer regime and determine the limiting T_g, α_g, and K_i as a function of $q_{cr\ cool}$ and D of the wall.

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